Field measurements and simulations of CO2 refrigerant system for supermarket

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## Nomenclature

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFC</td>
<td>Cholorofluocarbure</td>
</tr>
<tr>
<td>CO2</td>
<td>carbon dioxide</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient Of Performance</td>
</tr>
<tr>
<td>Dh</td>
<td>difference of enthalpy</td>
</tr>
<tr>
<td>dT</td>
<td>difference of temperature</td>
</tr>
<tr>
<td>$\dot{E}$</td>
<td>electrical power</td>
</tr>
<tr>
<td>FA</td>
<td>low temperature unit</td>
</tr>
<tr>
<td>GWP</td>
<td>Global Warming Potential</td>
</tr>
<tr>
<td>$\eta$</td>
<td>efficiency</td>
</tr>
<tr>
<td>HCFC</td>
<td>Hydrochlorofluocarbure</td>
</tr>
<tr>
<td>HFC</td>
<td>Hydrofluorocarbure</td>
</tr>
<tr>
<td>KA</td>
<td>low temperature unit for referent system; medium temperature unit for CO2 system</td>
</tr>
<tr>
<td>LR</td>
<td>Load Ratio</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>mass flow</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone Depletion Potential</td>
</tr>
<tr>
<td>Q</td>
<td>cooling capacity</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density</td>
</tr>
<tr>
<td>RS</td>
<td>Referent system</td>
</tr>
<tr>
<td>TR</td>
<td>Transcritical system</td>
</tr>
<tr>
<td>$\dot{V}$</td>
<td>swept volume</td>
</tr>
<tr>
<td>VKA</td>
<td>medium temperature unit</td>
</tr>
</tbody>
</table>

### Subscript:

- abs: absolute
- booster: booster system
- chiller: medium temperature
- comp: compressor
- corr: correction
- el: electric
- evap: evaporator
- freezer: low temperature
- in: inlet
- v: volumetric
1. **Introduction**

**Background**

The choice of a refrigerant in the supermarket refrigeration system depend this efficiency but also its environmental impact since the Montreal Protocol in 1987 and the Kyoto Protocol in 1995.

CFCs have the highest potential toward the depleting ozone layer. Hence since the Montreal Protocol the using of this kind of refrigerant are forbidden. CFCs have replaced by HCFCs refrigerant; theses refrigerants contribute also to the depleting ozone layer and the global warming. Based on this problem of ozone layer and global warning, HFCs have been created in order to replace CFCs and HCFCs. The Ozone Depletion Potential of HFC is null however the Global Warming Potential is already high. Therefore HFCs has integrated the Kyoto Protocol in 2005.

HFCs have an Ozone Depletion Potential equal to zero the reference is the R22 with an Ozone Depletion Potential equal to one. The Global Warming Potential is a problem for this refrigerant. For example the R404A, which is the mainly HFCs used, have a Global Warming Potential equal to 3800. That is to say its greenhouse impact is 3800 times important than the CO$_2$ which is the reference. Therefore some restrictive measures are limiting the amount of HFCs in the refrigeration system and mandatory leak detections. The finality is to stop the using of HFCs.

The first refrigerant used was natural. Since the environmental issue created by man-made refrigerant, news research on the natural refrigerant begin back. The natural refrigerants are mainly the ammonia and the CO$_2$. In the refrigeration industry the CO$_2$ is the commonly used although its thermodynamic properties are not as good as ammonia. The main problem with the ammonia is the toxicity; indeed it is toxic at very low concentration contrary to the CO2.

The main difficulties for the natural refrigerant is to be at least efficient than the man-made refrigerant such as HFCs. This is the case to be a serious candidate in new refrigeration systems.

**Energy using in Supermarket**

The main parts of interest from supermarket refrigeration are to be efficient, without environmental impact, and to be cost-efficient (cost of installation and maintenance). From the Figure 1-1 it is obvious that the main part of consumption is the refrigeration. Moreover after the air conditioning the refrigeration is the second to have the highest refrigerant loses of about 15 to 30% of the total charge. Supermarket is the second emitter of refrigerant in the atmosphere due to a leakage. These leakages are the main cause to the ozone layer depletion and the global warming.

Recently, the main refrigerants used in supermarket refrigeration are CFC/HCFC blends. Due the Ozone Depletion Potential of CFC and HCFC, these ones are replaced by HFCs which have zero Ozone Depletion Potential. However HCF has a high Global Warming Potential. These environmental impacts have given ways for various innovations in supermarket refrigeration system.
2. **Objective**

The objectives are to analyse and evaluate the performance of five systems (three HFC-based systems and two CO\textsubscript{2}-based systems) in order to compare the CO\textsubscript{2}-based solution against conventional system.

HFC-based systems are call referent system, they are varying in terms of their load, their climatic conditions. Although in order to compare each other a fair calculation method should be established.

CO\textsubscript{2}-based systems are varying in terms of loads, climatic conditions but also in terms of configuration and control.

The objectives to carry out are:

- Collect information on this CO\textsubscript{2} and HFC-based systems; their configuration and their control.
- Recovery measured data of this five system
- Used last template to calculate the mains parameters and the performance of each systems.
- Check the results with the previous result of these systems.
- Comparing the five systems

**Project**

Usage of natural refrigerant become a subject of actuality in the cold using with the application of a new legislation in order to control the manufacturing and the using of synthetic refrigerant. Natural refrigerant could become a solution of the environmental issue if this new systems are enough efficient comparing the conventional system. The performance of the system depends of various parameters such as the climatic condition, the control strategy, the demand, the configuration...

Sveriges Energi-&Kylcentrum (SEK) wich is a subsidiary company of Installatörernas Utbildningscentrum (IUC) in Katrineholm initialized this project in order to analyze and evaluate the
application of CO₂-based technologies in supermarket with a focus on energy efficiency and environmental issues. In many CO₂ supermarket installations in Sweden analysis have not been carried out to study the performance of the systems. Previously investigations have been done on CO₂ supermarket system solution as a cooperation project between SEK and KTH which included computer simulation modeling and experimental alternative to conventional solution.

**Project Partner**

<table>
<thead>
<tr>
<th>Organization</th>
<th>Participant/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sveriges Energi- &amp; Kylcentrum</td>
<td>Jörgen Rogstam</td>
</tr>
<tr>
<td>KTH – Energiteknik</td>
<td>Björn Palm / Samer Sawalha</td>
</tr>
<tr>
<td>ICA</td>
<td>Per-Erik Jansson</td>
</tr>
<tr>
<td>Green and Cool</td>
<td>Micael Antonsson</td>
</tr>
<tr>
<td>Partor AB</td>
<td>Martin Johansson</td>
</tr>
<tr>
<td>WICA</td>
<td>Peter Rylender</td>
</tr>
<tr>
<td>Ashell</td>
<td>Torbjörn Larsson</td>
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<tr>
<td>Huurre</td>
<td>Göran Sundin</td>
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<td>AGA</td>
<td>Christer Hens</td>
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<td>Transter</td>
<td>Ulf Vestergen</td>
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<td>Cupori</td>
<td>David Sharp</td>
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<td>Oppunda Svets</td>
<td>Ken Johansson</td>
</tr>
<tr>
<td>Energimyndighenten</td>
<td>Conny Ryytty</td>
</tr>
</tbody>
</table>

3. **Refrigerant in supermarket**

**Requirement for refrigerant**

Few parameters are necessary for a fluid to be a refrigerant. Moreover one refrigerant can’t be use for different applications (example: acclimatization and cryogenic).

Four classes can be notice:

- **Chemical**: It has to be stable and inert.
- **Health, safety and environmental**: Non-toxic, Non-flammable, low ODP and low GWP.
- **Thermal**: Low vapor heat capacity, low viscosity, high thermal conductivity, and critical point and boiling point appropriate for application.
• **Miscellaneous**: oil miscibility, low freezing point, easy leak detection, reasonable containment materials and low cost. (al., 2009)

The most important point is the chemical stability, in order to avoid chemical reaction with container material (tank and pipe). Since the Montreal’s agreement, companies have to choice a refrigerant with a low ODP.

In writing bellow, few made-man refrigerants compare to two natural refrigerants:

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Group</th>
<th>Ozone Depletion Potential</th>
<th>Global Warming Potential</th>
</tr>
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<tbody>
<tr>
<td>R-11</td>
<td>CFC</td>
<td>1</td>
<td>1800</td>
</tr>
<tr>
<td>R-12</td>
<td>CFC</td>
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<tr>
<td>R-113</td>
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<td>R-114</td>
<td>CFC</td>
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<td>9800</td>
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<tr>
<td>R-115</td>
<td>CFC</td>
<td>0.6</td>
<td>7200</td>
</tr>
<tr>
<td>R-502</td>
<td>CFC/HCFC blend</td>
<td>0.33</td>
<td>4300</td>
</tr>
<tr>
<td>R-22</td>
<td>HCFC</td>
<td>0.05</td>
<td>1700</td>
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<tr>
<td>R-123</td>
<td>HCFC</td>
<td>0.02</td>
<td>90</td>
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<td>R-32</td>
<td>HFC</td>
<td>0</td>
<td>675</td>
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<tr>
<td>R-125</td>
<td>HFC</td>
<td>0</td>
<td>2800</td>
</tr>
<tr>
<td>R-134a</td>
<td>HFC</td>
<td>0</td>
<td>1430</td>
</tr>
<tr>
<td>R-407C</td>
<td>HFC blend</td>
<td>0</td>
<td>1610</td>
</tr>
<tr>
<td>R-404a</td>
<td>HFC blend</td>
<td>0</td>
<td>3900</td>
</tr>
<tr>
<td>R-744 (CO2)</td>
<td>Natural</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>R717 (Ammonia)</td>
<td>Natural</td>
<td>0</td>
<td>0</td>
</tr>
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</table>

**Figure 3-1: List of few refrigerants with their ODP and GWP**

**CO2 as refrigerant**

CO2 as a refrigerant has used since the XIX century in order to produce cold temperature in the fishing boat. It has forgotten after the using of synthetic molecular CFCs, HCFCs and HFCs. However this new synthetic refrigerant has bad consequences over the environment. Indeed synthetic refrigerants have a global warming potential 1000 to 9000 times more important than the CO2 for CFCs and around 2000 times for HFCs.

**General properties of carbon dioxide**

**Heat exchange and high pressure compression**

**Efficiency of CO2 versus synthetic refrigerants**

4. **System description**

This part gives a general description of each system study here. There is two different systems, the CO2-based system (TR1 and TR2) and the conventional system use like a reference (RS1, 2 and 3).
Referent system
The 3 systems get the same schema. An indirect system on the medium temperature side and a direct system on the low temperature side with a sub cooling:

Supermarket refrigeration system RS1
RS1 is located in the northern part of Sweden, is open since October 2008. It is composed of one medium temperature stage (VKA1) and a low temperature stage (KA1). The secondary circuit on the evaporator are connected to a single propylene glycol circuit.

Both medium and low temperature stage use R404A as a refrigerant. Both medium and low temperature stage incorporates a sub-cooler after the condenser in order to sub-cooling the refrigerant before come in the evaporator. An internal heat exchanger is connected to the medium temperature and the low temperature stage to further sub-cool the refrigerant coming out of the sub-cooling.

An electronic valve is used on the medium temperature side while a thermostatic expansion valve is used on the low temperature valve.

Two frequency controlled compressor operate in tandem on both medium and low temperature stages.

The maximum design cooling capacity of the compressor on the medium temperature side is 87kW and 18kW for the low temperature side. The maximum demand for the room's and display is 70kW at the medium temperature side and 18kW for the low temperature side.
The system design provider, Partor AB, provided the information that the power consumed by the pump is not measured but knows it run at a constant speed, so it consumed the same energy during the year. The power used by the pumps it assumed constant, hence for RS1, 1.5kW for the pump on the low temperature side and 1.5kW for the pump on the medium temperature side.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Specification</th>
</tr>
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<tr>
<td>System specification</td>
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<tr>
<td>Sub-cooler</td>
<td>Yes</td>
</tr>
<tr>
<td>Heat recovery</td>
<td>NO</td>
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<tr>
<td>Refrigerant used</td>
<td>Primary refrigerant R404A (VKA and KA)</td>
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<td></td>
<td>Secondary refrigerant Propylene</td>
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<td>Compressor specification</td>
<td>Medium temperature stage Semi-hermetic reciprocating compressor, Bitzer 4J-22.2 (Tandem)</td>
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<td>Low temperature stage Semi-hermetic reciprocating compressor, Bitzer 4VCS-6.2 (Tandem)</td>
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<td>Heat exchanger specification</td>
<td>Internal heat exchanger Plate heat exchanger</td>
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<td></td>
<td>Evaporator Plate heat exchanger</td>
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<td></td>
<td>Condenser Plate heat exchanger</td>
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<tr>
<td>Expansion valve expansion</td>
<td>Medium temperature stage Electronic</td>
</tr>
<tr>
<td></td>
<td>Low temperature stage Thermostatic</td>
</tr>
<tr>
<td>Pump consumption</td>
<td>Medium temperature stage 1.5kW</td>
</tr>
<tr>
<td></td>
<td>Low temperature stage 1.5kW</td>
</tr>
</tbody>
</table>

Table 4-1: Major system details of RS1

Supermarket refrigeration RS2

RS2 is located near Tumba, is open since October 2008. It incorporate two medium temperature and two low temperature units (VKA1&2 and KA1&2). The secondary circuit is connected on the evaporator and condenser to the ethylene glycol circuit.

R407C is used as refrigerant on the medium temperature stage while R404A is used as refrigerant on the low temperature stage. Both medium and low temperature stage incorporates a sub-cooler after the condenser in order to sub-cooling the refrigerant before come in the evaporator. An internal heat exchanger is connected to the medium temperature and the low temperature stage to further sub-cool the refrigerant coming out of the sub-cooling.

Two frequency controlled compressors working in tandem on the medium temperature side (VKA1&2) while a single frequency controlled compressor is used on the low temperature side (KA1&2).

The system design provider, Partor AB, provided the information that the power consumed by the pump is not measured but knows it run at a constant speed, so it consumed the same energy during the year. The power used by the pumps it assumed constant, hence for RS1, 3kW for the pump on the low temperature side and 3kW for the pump on the medium temperature side.
### Parameters Specification

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<td></td>
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<td></td>
<td>Heat recovery: No</td>
</tr>
<tr>
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<td>Primary refrigerant: R404A (KA) and R407C (VKA)</td>
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<tr>
<td></td>
<td>Secondary refrigerant: Propylene</td>
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<td>Compressor specification</td>
<td>Medium temperature stage: Semi-hermetic reciprocating compressor, Bitzer 4J-22.2 (Tandem)</td>
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<td></td>
<td>Low temperature stage: Semi-hermetic reciprocating compressor, Bitzer 4J13.2 (Single)</td>
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<tr>
<td>Heat exchanger specification</td>
<td>Internal heat exchanger: Plate heat exchanger</td>
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<td></td>
<td>Evaporator: Plate heat exchanger</td>
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<tr>
<td></td>
<td>Condenser: Plate heat exchanger</td>
</tr>
<tr>
<td>Expansion valve expansion</td>
<td>Medium temperature stage: Electronic</td>
</tr>
<tr>
<td></td>
<td>Low temperature stage: Thermostatic</td>
</tr>
<tr>
<td>Pump consumption</td>
<td>Medium temperature stage: 3kW</td>
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<td></td>
<td>Low temperature stage: 3kW</td>
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</table>

**Tableau 4-2: Major system details of RS2**

**Supermarket refrigeration system RS3**

RS3 is located to Birsta and is open since March 2008. It incorporates two medium temperature (VKA1&2) and two low temperature units (KA&2). Heat is recovered from the system VKA5 and the air conditioner VKA3 is also used during the summer period.

R404A is used on one of the medium temperature stage (VKA1) while the other one used as refrigerant R407C. For the low temperature stage R404A is used as refrigerant. Both medium and low temperature stage incorporates a sub-cooler after the condenser in order to sub-cooling the refrigerant before come in the evaporator. An internal heat exchanger is connected to the medium temperature and the low temperature stage to further sub-cool the refrigerant coming out of the sub-cooling.

Both low and medium temperature units use frequency controlled compressor working in tandem.

The system design provider, Partor AB, provided the information that the power consumed by the pump is not measured but knows it run at a constant speed, so it consumed the same energy during the year. The power used by the pumps it assumed constant, hence for RS1, 6kW for the pump on the low temperature side and 6kW for the pump on the medium temperature side.
**CO₂-based system**

There are two kind of transcritical system study here, TR1 with four standard separated units and TR2 with two boosters systems and one standard unit.

**Supermarket refrigeration system TR1**

TR1 is located in the northern part of Sweden to Haparanda, it open since autumn 2007. The maximum design cooling capacity of the compressor is 230kW for the medium temperature and 60kW for the low temperature. It incorporates four units; two medium temperature units and two low temperature units, with an indirect water-glycol system for the heat rejection. The nearest station to the supermarket is Störon.

---

**Table 4-3: Major system details of RS3**

<table>
<thead>
<tr>
<th>Heat exchanger specification</th>
<th>Low temperature stage</th>
<th>Semi-hermetic reciprocating compressor, Bitzer 4H-15.2 (Tandem)</th>
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<td>Electronic</td>
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<td></td>
<td>Low temperature stage</td>
<td>Thermostatic</td>
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<td>Pump consumption</td>
<td>Medium temperature stage</td>
<td>6kW</td>
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<tr>
<td></td>
<td>Low temperature stage</td>
<td>6kW</td>
</tr>
</tbody>
</table>

---

**Figure 4-2: simplified circuit of TR1 refrigeration system**
There are 4 compressors on the medium temperature side for each medium temperature units KA1 and KA2. There are 2 compressors on the low temperature side, with intercooler, for each low temperature units (double stage compression).

Since March 2009, there is a frequency converter on each compressor of FA2 and KA2, in order to reduce the power consumption. In the practice there isn’t a frequency measurement on Iwmac website; so the swept volume is deduce by the power consumption. And with this swept volume it is possible to calculate the mass flow in the system.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>System specification</td>
<td></td>
</tr>
<tr>
<td>System</td>
<td>Direct/ Indirect for heat rejection</td>
</tr>
<tr>
<td>Oil cooler</td>
<td>Yes</td>
</tr>
<tr>
<td>Coolant</td>
<td>Yes</td>
</tr>
<tr>
<td>Refrigerant used</td>
<td>Primary refrigerant</td>
</tr>
<tr>
<td>Compressor specification</td>
<td></td>
</tr>
<tr>
<td>Medium temperature stage</td>
<td>Transcritical CO2, single-stage / Compressor Dorin TCS 373-D</td>
</tr>
<tr>
<td>Low temperature stage</td>
<td>Transcritical CO2, two stage with intercooler/ Compressor Dorin TCDH 372 B-D</td>
</tr>
</tbody>
</table>

Table 4-4: Major system details of TR1

**Supermarket refrigeration system TR2**

TR2 is located near Goteborg; it is open since August 2008. The maximal cabinet design cooling load is 200kW for the medium temperature and 50kW for the low temperature. It incorporates three separated units; two booster systems for the medium temperature and low temperature cabinets. The load ration for this booster system is around 2. The last unit is a standard transcritical system for medium temperature.
Figure 4-3: simplified circuit of TR2 refrigeration system

The sub-cooling before the evaporator comes from the ground. This source is connected to the two booster systems and the standard system.

For booster system, the electrical power consumption for the low temperature compressor and the all compressors is known only for KAFA1. For KAFA2 only the total electrical power consumption is known. In order to permit the calculation the assumption is the electrical power consumption ratio between the low temperature compressor and the total consumption is the same for KAFA1 and KAFA2.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>System specification</td>
<td>System</td>
</tr>
<tr>
<td></td>
<td>Direct/ Indirect for heat rejection</td>
</tr>
<tr>
<td>Oil cooler</td>
<td>Yes</td>
</tr>
<tr>
<td>Heat recovery</td>
<td>Yes</td>
</tr>
<tr>
<td>Subcooling</td>
<td>Ground heat sink</td>
</tr>
<tr>
<td>Refrigerant used</td>
<td>Primary refrigerant</td>
</tr>
<tr>
<td></td>
<td>R477</td>
</tr>
<tr>
<td>Booster unit’s compressors</td>
<td>Medium temperature stage</td>
</tr>
<tr>
<td></td>
<td>Transcritical CO2 / Compressor Dorin SCS 362</td>
</tr>
<tr>
<td>Low temperature stage</td>
<td>Transcritical CO2, / Compressor Dorin TCS 373</td>
</tr>
<tr>
<td>Standard unit’s compressor</td>
<td>Medium temperature</td>
</tr>
<tr>
<td></td>
<td>Transcritical CO2, / Compressor Dorin TCS 373</td>
</tr>
</tbody>
</table>

Tableau 4-5: Major system details of TR2
5. **Measurements and evaluation methods**

For the evaluation of the coefficient of performance, the cooling capacity, the knowledge of the fluid’s temperature and pressure is necessary. Pressure and temperature permit to calculate the enthalpy and with the mass flow to define the cooling capacity and others losses. The mass flow rate is not measured directly but evaluate from the compressor side with the constructor’s data. The COP is calculating using the cooling capacity and the measured or calculated electrical consumption.

**Pressure and temperature measurement**

Data recovery of the refrigerant thermodynamics states are the pressure and the temperature. The temperature sensor types are PT100 or PT1000 widely used in the refrigeration regulation. The pressure sensors give absolute or relative pressure depending their initial condition.

**CO₂-based system**

For CO₂-based systems the electrical consumption is directly given. On contrary referent systems the electrical consumption has to be calculated by the different refrigerant’s thermodynamic states near the compressor.

The sensors are not installed especially for this study but in order to control and regulate the system.

During this study, the data’s synchronization of CO₂-based system was wrong. Using of a small algorithm permitted to solve this problem but with less accuracy. The time interval between two values is 10 or 20 minutes.

Below an example of Iwmac’s interface use to recovery parameters:
The main problem occurred was the synchronization of data. In order to solve it a small program has written in Python; below its interface:

![Iwmac interface. Example of Hovås KAFA1 booster units](image)

**Figure 5-1:** Iwmac interface. Example of Hovås KAFA1 booster units

It works with an average method between two dates separate by a defined step time; for the two CO₂-based systems the step time was 10 minutes.

**Referent system**

For referent system, the software used was different. The synchronization was good with 15 minutes interval.
The condensing temperature corresponds to the bubble temperature for the condensing pressure. And the evaporating temperature corresponds to the dew temperature for the evaporating pressure. For refrigerants R407C with a large glide temperature during the condensing, the condensing temperature is assumed as the bubble temperature plus two degrees.

**Electrical power consumption: measured or calculated**

In the case of the system is equipped of sensor consumption, the electrical power is directly recovery in the data. This power includes the compressor, the pumps and fans consumption. This king of value is used in CO₂-based system.

However, when the field of measurement doesn’t incorporate sensor consumption, the electrical consumption for the compressor is calculated with the difference of the thermodynamic fluid state over the compressor. It is the method used for referent system. The loose for the compressor are assumed around 7% of the total consumption:

\[ \dot{E}_{el} = \dot{m} \times \Delta h_{comp} \times 1.07 \]

Where,

\( \Delta h_{comp} \) is the enthalpy difference over the compressor

\( \dot{m} \) is the mass flow

**The cooling capacity**

The cooling capacity (\( \dot{Q} \)) of the system at the evaporator calculated with the mass flow and the difference of fluid thermodynamic state over the evaporator:

\[ \dot{Q} = \dot{m} \times \Delta h_{evap} \]
Where,

\( \Delta h_{\text{evap}} \) is the enthalpy difference over the evaporator

\( \dot{m} \) is the mass flow

The enthalpy difference over the evaporator is the difference between the enthalpy at the sub cooler and the enthalpy at compressor inlet.

**Mass flow evaluation**

The mass flow is necessary to calculate the cooling capacity and system's looses. None of our supermarket used equipment permitting to obtain directly this value. So a calculation method is used. This method is based on the refrigerant's pressure and the temperature at the compressor inlet to get the specific volume. The compressor data is necessary to know the swept volume and the volumetric efficiency. The mass flow is given by this equation:

\[
\dot{m} = \frac{\eta_v \cdot \dot{V}}{\nu_{\text{comp_in}}} \quad [\text{kg/s}]
\]

\( \eta_v \) = Volumetric efficiency [-] based on the compressor data

\( \dot{V} \) = swept volume [m\(^3\)/s] based on the compressor data

\( \nu_{\text{comp_in}} \) = specific volume [m\(^3\)/kg] \( \rightarrow \) \( f_{\text{state}}(P_{\text{abs,comp_in}}, T_{\text{comp_in}}) \)

In the case where a frequency converter is used, the previous equation has to integrate this information. So the mass flow can be calculated by:

\[
\dot{m} = \eta_v \times \dot{V} \times \rho \times \frac{\Delta h}{50} \quad [\text{kg/s}]
\]

\( \eta_v \) = Volumetric efficiency [-] based on the compressor data

\( \dot{V} \) = swept volume [m\(^3\)/s] based on the compressor data

\( \rho \) = density [kg/m\(^3\)] \( \rightarrow \) \( f_{\text{state}}(P_{\text{abs,comp_in}}, T_{\text{comp_in}}) \)

The swept volume is given indirectly in the constructor data. It depends of the compressor rotation speed and the volume inside the piston:

\[
\dot{V} = V \times \frac{\text{RPM}}{60} \quad [\text{m}^3/\text{s}]
\]

\( V \) = swept volume, inlet the compressor [m\(^3\)]

RPM = Rotation per minutes [#/min]
**CO₂-based system: volumetric efficiency compressor**

![Figure 5-4: Volumetric efficiency of CO₂ compressors](image1)

Referent system: volumetric efficiency compressor

![Figure 5-5: Volumetric efficiency of conventional compressors](image2)

**COP calculation**

In this study, the value the most interesting is the Coefficient Of Performance of the system. It is generally defined using this equation:

\[
COP = \frac{Q}{E_{el}}
\]

Where,
\( Q \) is the cooling capacity

\( E_{el} \) is the electrical consumption

In the electrical consumption is included the compressor consumption, the pumps consumption and the fans consumption.

**CO\(_2\)**-based system

![Diagram of TR1 refrigeration system]

*Figure 5-6: TR1 refrigeration system*

For TR1 all the systems are separate, so the COP is the ratio of the cooling capacity and the electrical consumption.
Figure 5-7: TR2 refrigeration system, with KAFA booster unit and KA standard medium temperature unit

For TR2, a COP for a booster system is calculated in a different way. In this case the high stage compressor and the booster compressor are located in different places in the system; it is possible to calculate two mass flows. One mass flow is the total mass flow going through the high stage compressors and one mass flow is the mass flow maintaining the freezers. In writing below the mass balance:

\[ m_{	ext{chiller}} = m_{	ext{total}} - m_{	ext{freezer}} \]

The total COP of the system:

\[ COP_{	ext{tot}} = \frac{Q_{	ext{freezer}} + Q_{	ext{chiller}}}{E_{	ext{freezer}} + E_{	ext{chiller}} + E_{	ext{pumps}}} \]

In order to calculate the COP of the booster, only the cooling capacity from the freezer side and the capacity from the medium temperature side which goes to the medium temperature cabinets is taken into account. The medium temperature power used for the condensation on the freezer side is eliminated.

\[ COP_{	ext{booster}} = \frac{Q_{	ext{freezer}} + Q_{	ext{chiller}} - Q_{c, \text{freezer}}}{E_{	ext{freezer}} + E_{\text{chiller}}} \]

The condenser load of the CO₂ unit can be calculated with:
\[ \dot{Q}_{c,\text{freezer}} = \dot{Q}_{\text{freezer}} + \dot{E}_{\text{comp.shaf.freezer}} \]

And to decide the load of the medium temperature side cabinet:

\[ \dot{Q}_{\text{cab}} = \dot{Q}_{\text{chiller}} - \dot{Q}_{c,\text{freezer}} \]

The electrical energy from the chillers which goes to the freezer is calculated by:

\[ \dot{E}_{\text{chiller.for.freezer}} = \frac{\dot{Q}_{\text{freezer}}}{\dot{Q}_{\text{high}}} \times \dot{E}_{\text{chiller}} \]

The COP for the freezer can be calculated by:

\[ COP_{\text{freezer}} = \frac{\dot{Q}_{\text{freezer}}}{\dot{E}_{\text{freezer}} + \dot{E}_{\text{chiller.for.freezer}} + \dot{E}_{\text{pumps}}} \]

The COP for the chiller can be calculated by:

\[ COP_{\text{chiller}} = \frac{\dot{q}_{\text{cab}}}{\dot{E}_{\text{chiller}} + \dot{E}_{\text{chiller.for.freezer}} + \dot{E}_{\text{pumps}}} \]

In order to compare all system each other, the load ratio has to be identical. The load ratio is the ratio between the cooling capacity from the chiller and the cooling capacity from the freezer. In Europe the load ratio is around 3 so 3 times more cooling capacity from the chiller than the freezer. In order to correct the COP, using a fix load ratio (LR_{corr}) like writing below:

\[ COP_{\text{tot.LR}} = \frac{\dot{q}_{\text{chiller}} \times (1 + LR_{corr})}{\frac{1}{LR_{corr}} \times \dot{E}_{\text{freezer}} \times \dot{q}_{\text{chiller}} + \dot{E}_{\text{chiller}}} \]
Reference system

![Reference refrigeration system diagram]

The coefficient of performance for the medium temperature side is written below:

$$\text{COP}_{\text{chiller}} = \frac{Q_{\text{chiller}}}{\dot{E}_{\text{el, chiller}} + \dot{E}_{\text{pumps}}}$$

For the low temperature side, a part of the power from the medium temperature side is used in order to subcool the fluid in the low temperature side.

$$\text{COP}_{\text{freezer}} = \frac{Q_{\text{chiller}}}{\dot{E}_{\text{el, freezer}} + (\text{subcooler}) \text{COP}_{\text{chiller}}}$$

The electrical power from the medium temperature stage is used to subcool the low temperature side:

$$\dot{E}_{\text{med to freezer}} = \frac{Q_{\text{subcooler}}}{\text{COP}_{\text{chiller}}}$$

Finally, the total coefficient of performance of the system incorporates the chiller, the freezer and the brine pumps:

$$\text{COP}_{\text{tot}} = \frac{Q_{\text{chiller}} + Q_{\text{freezer}}}{\dot{E}_{\text{el, chiller}} + \dot{E}_{\text{el, freezer}} + \dot{E}_{\text{pumps}}}$$
6. **General system performance analysis**

This part includes an analysis of each system described above, with few graphics showing the behaviour during the observation period. Mainly it shows the cooling capacity, the electrical power consumption and the Coefficient Of Performance.

**Analysis of TR1**

The [Figure 6-1](#) shows the cooling capacity for the low temperature (FA1&2) and the medium temperature side (KA1&2). The medium temperature is around -10°C and the low temperature around -35°C; these values didn’t change since last year.

![Cooling capacity graph](#)

*Figure 6-1: Cooling capacity on the medium temperature side and on the low temperature side during the observation period.*

Peak de consumption appears during the summer period when the outdoor temperature increase. This is almost visible on the medium temperature side. However the outdoor temperature has less effect on the low temperature side because the cabinets are protected by a glass doors. These glass doors decrease the exchange with the ambient and loosess.

The [Figure 6-2](#) shows the electrical power consumption during the observation period. The curves follow the same trend as the cooling capacity.
Figure 6-2: Electrical consumption for medium temperature side and for the low temperature side during the observation period.

The coolant temperature is influenced by the outdoor temperature. There is a correlation between the COP and the coolant temperature. The data recovered show clearly this correlation (Figure 6-3).

Figure 6-3: COP function of the coolant temperature inlet the gas cooler for the medium temperature and the low temperature side
The performance of the system decrease when the coolant temperature increases. The reason is because the high pressure increases so the pressure ratio is higher. When the pressure ratio is higher the electrical consumption is more important because the efficiency of the compressor decrease when the pressure ration increase.

**Analysis of TR2**

For this system, sub cooling have an important influence; the heat recovery, using a heat pump, is used during the winter in order to heat the supermarket. The Figure 6-4 shows the main characteristics values of KAFA1 unit; it means it cooling capacity, the electrical power consumption, the condensing temperature, the outdoor temperature and the effect of sub cooling borehole:

![Figure 6-4: Mains parameters of KAFA1 booster unit during the observation period.](image)

The sub cooling is expressed by a differential of temperature (\(\Delta T\)) in Kelvin.

During the winter, the cooling capacity drop, as expected. However the electrical consumption is almost the same, although normally this power should decrease du the low outdoor temperature. The reason is during the winter the condensing temperature is maintained to 25°C in order to increase the capacity of the heat recovery system. The refrigeration system and the heat pump are connected via the borehole but also on the “warm” side through a plate heat exchanger disposed on the high pressure circuit at the compressor exit. It shows on the plot, the differential temperature of the sub cooling borehole is more important during the winter than the summer.

The second using of the borehole is to subcool the fluid in order to maintain the COP to a high level even if the condensing temperature increases. Moreover the subcooling differential temperature is more important when the condensing temperature is high.
This method to connect a heat pump to the subcooling of the refrigeration system explains the behavior of system performance. The same behavior is plot in the next figure for KAFA2 (Figure 6-5) and KA3 (Figure 6-6).

![Diagram](image1)

*Figure 6-5: Mains parameters of KAFA2 booster unit during the observation period.*

The observations are the same as the previous booster unit KAFA1. However for KAFA2 the cooling capacity has been the almost same and lower than KAFA1 during the observation period. And the electrical power consumption is also lower.

![Diagram](image2)

*Figure 6-6: Mains parameters of KA3 standard unit during the observation period.*
The KA3 unit confirms the previous observation over the cooling capacity and the heat recovery.

The Figure 6-7 shows the trend of the COP of each unit during the observation period. The COP decreases during the winter due to the high condensing temperature used for the heat recovery system. The subcooling seems not compensate totally these losses. The difference between the two booster units is may be due to a missing parameter over the electrical consumption on the medium temperature part. This assumption is the two units have the same electrical consumption ratio at the same time. It was the only assumption possible in order to purchase the calculation of KAFA2.

As can be seen on the last figure, the COP of the boosters units are between 2 and 2.5. For the medium temperature unit the COP is around 4.

![COP Graph](image)

Figure 6-7: COP for each unit during the observation period.

**Analysis of RS1**

One of the most important parameters in refrigeration is the cooling capacity. The Figure 6-8 shows the mains parameters of RS1 for the medium temperature unit during the observation period. This plot shows the cooling capacity, the electrical power consumption, the evaporating temperature, and the outdoor temperature. The cooling capacity depends on the electrical consumption and is influence by the outdoor temperature.
Figure 6-8: Mains parameters for the medium temperature side during the observation period

The Figure 6-9 shows the cooling capacity, the electrical consumption, the evaporating temperature, and the outdoor temperature during one year. The evaporating temperature is almost constant during the year it means -8°C for the chillers and -29°C for the freezers.

Figure 6-9: Mains parameters for the low temperature side during the observation period

The cooling capacity on the medium temperature side has more fluctuation than the low temperature side. The reason is the low temperature cabinets have a glass doors which decrease the heat exchange with the ambient. The fluctuation of the cooling capacity for the medium temperature
side is around 25kW (let 28% of the maximum design cooling capacity) although for the low medium side it is around 3kW (let 17% of the maximum design cooling capacity).

A peak of electrical consumption appears during the summer period in order to provide enough cooling power. The electrical consumption decreases during the winter. The ambient temperature has an influence on the condensing temperature and so has an influence on the compressor power consumption.

So the condensing temperature has an influence on the performance of the system. The condensing temperature for the medium temperature side is around 22°C while for the low temperature side the condensing temperature is around 16°C. Indeed, the Figure 6-10 shows, the differential of temperature between the condensing temperature and the outdoor temperature is one time and half important on the chiller that on the freezers. The condensing temperature for the chiller is a bit too high but it is a design default. The one way valve which located on the compressor discharge is sized very small hence causing a higher pressure drop in the compressor discharge line which in turn leads to a higher discharge pressure than necessary.

![Figure 6-10: Condensing temperature for the medium and low temperature side, outdoor temperature, and the differential of temperature between the condensing temperature and the outdoor temperature.](image)

The Figure 6-11 shows the COP trend during the year. It appears clearly the COP decrease when the outdoor temperature increases, which increase the condensing temperature of the system. The high condensing temperature is the consequence of the low COP.
Analysis of RS2

RS2 use two different refrigerants, R404A for the freezers and R407C for chillers. The Figure 6-12 shows the monthly average of the condensing temperature for each unit (low and medium temperature unit) with the outdoor temperature during the observation period. It appears the condensing temperature is maintained to a same value for low and medium temperature units. However the unit VKA1 has always the highest condensing temperature.
The Figure 6-13 shows the cooling capacity and the electrical power consumption of RS2. The lowest cooling capacity appears during the winter when the outdoor temperature is the lowest. However the varying of cooling capacity is higher for the medium temperature units than the low temperature units. The reason is the same as for RS1: the cabinets of the low temperature units are protected by a glass door which decreases the exchange with the ambient. On the contrary the highest cooling capacity appears during the summer period when the outdoor temperature is the highest.

The electrical power consumption follows the same trend like the cooling capacity but with less variation. The electrical consumption increases in summer and decrease in winter.

The Figure 6-14 shows the subcooling capacity during the year. The evaporating temperature is maintained at a constant value it means -8°C for the medium temperature side and -32°C for the low temperature side. It can be noticed the subcooling capacity is the highest during the summer and the lowest during the winter. The trend of the subcooling capacity follows the trend of the outdoor temperature.
The Figure 6-15 shows the evolution of the COPs during the year for RS2. The highest COP is during the winter; it is logical because the demand is the lowest and the condensing temperature is reduced with the low outdoor temperature. On the contrary the COP is the lowest during the summer when the cooling demand is the higher and the outdoor temperature doesn’t permit a low condensing temperature. So the maximum COP for the chiller is 4.7 and for the freezer is 3.
Analysis of RS3

For RS3, the refrigerant R404A is used for the low temperature side and for one of the medium temperature unit (VKA1). The refrigerant R407C is used for the unit VKA2.

The Figure 6-16 shows the cooling capacity and the electrical consumption for the low and the medium temperature units. The same observation can be applied about the cooling capacity and the electrical consumption; it means a higher cooling capacity during the summer period than during the winter period. The cooling capacity variation is less important on the low temperature side because the cabinets have a glass door which decreases the heat exchange with the ambient.

Figure 6-16: Cooling capacity and electrical consumption for the medium and the low temperature units.

The Figure 6-17 shows the subcooling capacity of the freezers units. The evaporating temperature is almost constant for the medium and the low temperature side. The evaporating temperature for the medium temperature units is -7. The evaporating temperature for the low temperature unit is -31°C.

The subcooling capacity decreases for the winter period when the outdoor temperature is low and so when the condensing temperature decreases too.
7. **Comparison each system with previous result**

**CO2-based system**

Previously David Frelechox did the same calculation for Hovås and Haparanda supermarket from September 2008 to June 2009. In order to check the new result, it has been compared with previous results.

The comparison is applied on four parameters:

- The condensing temperature
- The evaporating temperature
- The COP function of the condensing temperature
- The COP function of the outdoor temperature

**TR1 system**

TR1 supermarket is compounded of four separate units: two medium temperature units and two low temperature units.

- KA1 unit

As show the **Table 7-1** the condensing and the evaporating temperature is almost the same.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009 results</th>
<th>2010 results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature</td>
<td>14.3°C</td>
<td>14.3°C</td>
</tr>
<tr>
<td>Evaporating temperature</td>
<td>-8.2°C</td>
<td>-7.8°C</td>
</tr>
</tbody>
</table>

*Table 7-1: Comparison of the condensing and the evaporating temperature for KA1*
The Figure 7-1 shows the comparison of the news and the olds results with the COP function of the condensing and the outdoor temperature. The 2010’s COPs are a bit higher than the olds ones, it can be explained by the evaporating temperature is higher. However the COP result is the same order, the news points fit the olds ones.

<table>
<thead>
<tr>
<th>COP function of the condensing temperature</th>
<th>COP function of the outdoor temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1" alt="Graph" /></td>
<td><img src="image2" alt="Graph" /></td>
</tr>
</tbody>
</table>

**Figure 7-1**: Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for KA1

➢ KA2 units

The Table 7-2 shows the condensing and the evaporating temperature is almost the same.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009 results</th>
<th>2010 results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature</td>
<td>15.5°C</td>
<td>15.3°C</td>
</tr>
<tr>
<td>Evaporating temperature</td>
<td>-8.3°C</td>
<td>-7.8°C</td>
</tr>
</tbody>
</table>

**Table 7-2**: Comparison of the condensing and the evaporating temperature for KA2

<table>
<thead>
<tr>
<th>COP function of the condensing temperature</th>
<th>COP function of the outdoor temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image3" alt="Graph" /></td>
<td><img src="image4" alt="Graph" /></td>
</tr>
</tbody>
</table>

**Figure 7-2**: Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for KA2
The Figure 7-2 shows the comparison of the COP function of the condensing and the outdoor temperature for the news results and the olds ones. The same observation can be made because the COP is a bit higher.

- FA1 unit

The Table 7-1 shows the condensing and the evaporating temperature is almost the same.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009 results</th>
<th>2010 results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temp.</td>
<td>15.3°C</td>
<td>14.9°C</td>
</tr>
<tr>
<td>Evap. temp.</td>
<td>-34.1°C</td>
<td>-35°C</td>
</tr>
</tbody>
</table>

Table 7-3: Comparison of the condensing and the evaporating temperature for FA1

![COP function of the condensing temperature](image1)

The Figure 7-3 shows the comparison of COP function the condensing and the outdoor temperature. The news points fit the olds ones.

- FA2 unit

The Table 7-4 shows the condensing temperature is the same while the condensing temperature is 2 degrees lower the old result.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009 results</th>
<th>2010 results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temp.</td>
<td>14.9°C</td>
<td>14.5°C</td>
</tr>
<tr>
<td>Evap. temp.</td>
<td>-32.5°C</td>
<td>-34.6°C</td>
</tr>
</tbody>
</table>

Table 7-4: Comparison of the condensing and the evaporating temperature for FA2

The Figure 7-4 shows the comparison of the COP function of the condensing and the outdoor temperature. The news points fit with the olds ones.
TR2 system

TR2 system is compounded of two boosters systems and one standard unit.

- KAFA1 unit

The Table 7-5 shows the comparison of the condensing and the evaporating temperature. There are some differences on the condensing maybe cause by the outdoor temperature.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009 results</th>
<th>2010 results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature</td>
<td>21°C</td>
<td>18.6°C</td>
</tr>
<tr>
<td>Medium evaporating temperature</td>
<td>-9°C</td>
<td>-10.1°C</td>
</tr>
<tr>
<td>Low evaporating temperature</td>
<td>-35.3</td>
<td>-35.2</td>
</tr>
</tbody>
</table>

Table 7-5: Comparison of the condensing and the evaporating temperature for KAFA1

**Figure 7-4:** Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for FA2.

**Figure 7-5:** Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for KAFA1
The Figure 7-5 shows the comparison between the COP function of the condensing and the outdoor temperature. The news points fit the olds ones.

 KAFA2

The Table 7-6 shows the comparison of the condensing and the evaporating temperature.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009 results</th>
<th>2010 results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature</td>
<td>21.8°C</td>
<td>19.9°C</td>
</tr>
<tr>
<td>Medium evaporating temperature</td>
<td>-10.4°C</td>
<td>-11.2°C</td>
</tr>
<tr>
<td>Low evaporating temperature</td>
<td>-35.1</td>
<td>-35</td>
</tr>
</tbody>
</table>

Table 7-6: Comparison of the condensing and the evaporating temperature for KAFA2

![Figure 7-6: Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for KAFA2](image)

The Figure 7-6 shows the comparison between the COP function of the condensing and the outdoor temperature. The news points fit the olds ones.

 KA3 unit

The Table 7-7 shows the comparison of the condensing and the evaporating temperature.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009 results</th>
<th>2010 results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature</td>
<td>21.8°C</td>
<td>19.9°C</td>
</tr>
<tr>
<td>Medium evaporating temperature</td>
<td>-10.4°C</td>
<td>-11.2°C</td>
</tr>
</tbody>
</table>

Table 7-7: Comparison of the condensing and the evaporating temperature for KA3

The Figure 7-7 shows the comparison between the COP function of the condensing and the outdoor temperature. The news points fit the olds ones.
COP function of the condensing temperature | COP function of the outdoor temperature
---|---
![COP function graphs](image)

**COP function of the condensing temperature**
- COP 2010
- COP 2008-2009

**COP function of the outdoor temperature**
- COP 2010
- COP 2008-2009

**Figure 7-7:** Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for KA3.

**Referent system**
Previously Manickam Louis Tamilarasan did the same calculation for Arvidsjaur, Tumba and Birsta supermarket from October 2008 to June 2009. In order to check the new result, it has been compared with previous results.

The comparison is applied on four parameters:

- The condensing temperature
- The evaporating temperature
- The COP function of the condensing temperature
- The COP function of the outdoor temperature

**RS1 system**

- Mains temperatures

The **Table 7-8** shows the comparison of the condensing and the evaporating temperature.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009</th>
<th>2009-2010</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature (medium part)</td>
<td>18.7°C</td>
<td>19.3°C</td>
</tr>
<tr>
<td>Condensing temperature (low part)</td>
<td>12.3°C</td>
<td>13.2°C</td>
</tr>
<tr>
<td>Medium evaporating temperature</td>
<td>-8.2°C</td>
<td>-8.1°C</td>
</tr>
<tr>
<td>Low evaporating temperature</td>
<td>-28.8°C</td>
<td>-29</td>
</tr>
</tbody>
</table>

**Table 7-8:** Comparison of the condensing and the evaporating temperature for RS1

- COPs

The **Figure 7-8** shows the comparison between the COP function of the condensing and the outdoor temperature. The news points fit the olds ones.
The Table 7-9 shows the comparison of the condensing and the evaporating temperature.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009</th>
<th>2009-2010</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature (medium part)</td>
<td>19.4°C</td>
<td>18.3°C</td>
</tr>
<tr>
<td>Condensing temperature (low part)</td>
<td>16.8°C</td>
<td>17.8°C</td>
</tr>
<tr>
<td>Medium evaporating temperature</td>
<td>-8.3°C</td>
<td>-7.8°C</td>
</tr>
<tr>
<td>Low evaporating temperature</td>
<td>-32.6°C</td>
<td>-32</td>
</tr>
</tbody>
</table>

Table 7-9: Comparison of the condensing and the evaporating temperature for RS2
The Figure 7-9 shows the comparison between the COP function of the condensing and the outdoor temperature. The news points fit the olds ones.

<table>
<thead>
<tr>
<th>Medium stage</th>
<th>COP function of the condensing temperature</th>
<th>COP function of the outdoor temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td><img src="image1.png" alt="Graph" /></td>
<td><img src="image2.png" alt="Graph" /></td>
</tr>
<tr>
<td>Low stage</td>
<td><img src="image3.png" alt="Graph" /></td>
<td><img src="image4.png" alt="Graph" /></td>
</tr>
</tbody>
</table>

**Figure 7-9: Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for RS2.**

**RS3 system**

- Mains temperatures

The Table 7-8 shows the comparison of the condensing and the evaporating temperature.

The difference on the condensing temperature for the freezer can be explained because the old analyse is from March 2009 to August 2009 while the new one is from August 2009 to December 2009. So the outdoor temperature is not the same.
### COPs

The Figure 7-10 shows the comparison between the COP function of the condensing and the outdoor temperature. The news points fit the olds ones.

<table>
<thead>
<tr>
<th></th>
<th>2008-2009</th>
<th>2009-2010</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing temperature (medium part)</td>
<td>20.9°C</td>
<td>20.5°C</td>
</tr>
<tr>
<td>Condensing temperature (low part)</td>
<td>21.1°C</td>
<td>14.5°C</td>
</tr>
<tr>
<td>Medium evaporating temperature</td>
<td>-7.6°C</td>
<td>-7.7°C</td>
</tr>
<tr>
<td>Low evaporating temperature</td>
<td>-30°C</td>
<td>-31.2°C</td>
</tr>
</tbody>
</table>

Table 7-10: Comparison of the condensing and the evaporating temperature for RS3

---

**Medium stage**

- **COP function of the condensing temperature**
  - COP VKA 2008-2009
  - COP VKA 2009-2010

**Low stage**

- **COP function of the condensing temperature**
  - COP KA 2008-2009
  - COP KA 2009-2010

---

Figure 7-10: Comparison of old and new results with the COP function of the condensing temperature and the outdoor temperature for RS3.
Summary

Finally all system kept their values and parameters since last year. The only difference is the outdoor temperature; indeed the winter 2010 was colder than previously. So it appears a condensing temperature a bit lower. However the COP is almost the same in comparison with last year.

8. Comparison between TR1 TR2 RS1 RS2 and RS3

In order to make a fair comparison between each parameter, these have to be compared with the same conditions of load ratio, and climatic condition. In the section 5 “COP evaluation” a formula is given in order to fix a load ration and calculated the COP with this one. For the climatic condition, for each system the COP is plot in function of the outdoor temperature and in function of the condensing temperature.

General characteristic of each system
The Table 8-1 shows the mains characteristics of refrigerants and temperature for each parameter.

<table>
<thead>
<tr>
<th></th>
<th>Temperature unit</th>
<th>Refrigerant</th>
<th>Condensing temperature [°C]</th>
<th>Evaporating temperature [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>TR1</td>
<td>Medium</td>
<td>CO2</td>
<td>14,8</td>
<td>-7,9</td>
</tr>
<tr>
<td></td>
<td>Low</td>
<td></td>
<td>14,7</td>
<td>-34,5</td>
</tr>
<tr>
<td>TR2</td>
<td>Medium</td>
<td>CO2</td>
<td>20,2</td>
<td>-10,6</td>
</tr>
<tr>
<td></td>
<td>Low</td>
<td></td>
<td>20,2</td>
<td>-35</td>
</tr>
<tr>
<td>RS1</td>
<td>Medium</td>
<td>R404A</td>
<td>19,3</td>
<td>-8,1</td>
</tr>
<tr>
<td></td>
<td>Low</td>
<td></td>
<td>13,2</td>
<td>-29</td>
</tr>
<tr>
<td>RS2</td>
<td>Medium</td>
<td>R407C</td>
<td>18,3</td>
<td>-7,8</td>
</tr>
<tr>
<td></td>
<td>Low</td>
<td>R404A</td>
<td>17,8</td>
<td>-32</td>
</tr>
<tr>
<td>RS3</td>
<td>Medium</td>
<td>R404A(VKA1);R407C(VKA2)</td>
<td>20,5</td>
<td>-7,7</td>
</tr>
<tr>
<td></td>
<td>Low</td>
<td>R404A</td>
<td>14,4</td>
<td>-31,2</td>
</tr>
</tbody>
</table>

Table 8-1: Summary of the mains characteristics of each system during their observation period.

It appears the evaporating temperature for the CO2-based system is often lower than the referent system. It is not a characteristic of this system but just a choice of the company to fixe this temperature to supply their cabinets.

The condensing temperature is a consequence of the geographic localisation of the supermarket. Indeed on the Northern part of Sweden the outdoor temperature is lower than on the South. The Figure 8-1 shows the distribution of these systems in Sweden.
Figure 8-1: Map of Sweden with the localisation of each system

**The load ratio**

Each system has its own cooling capacity but also its own load ratio. The load ratio is the ration between the cooling capacity from the chiller and the cooling capacity from the freezer.

<table>
<thead>
<tr>
<th></th>
<th>Average of the load ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>TR1</td>
<td>3,1</td>
</tr>
<tr>
<td>TR2</td>
<td>3,4</td>
</tr>
<tr>
<td>RS1</td>
<td>4,9</td>
</tr>
<tr>
<td>RS2</td>
<td>3,2</td>
</tr>
<tr>
<td>RS3</td>
<td>4,1</td>
</tr>
</tbody>
</table>

Table 8-2: List of Load Ration design for each system.
The Table 8-2 shows the load ratio design for each supermarket. These values are often upper three. Thereafter the fix load ratio is applied in order to permit to compare the systems each them. In this case the design load ration is upper 3 so the Coefficient Of Performance with a fix load ration of 3 will be lower than the real COP.

In writing below Figure 8-2 shows the total COP with their own design load ratio function of the time.

![Graph showing COP values for different systems]

Figure 8-2: Monthly average of the total COP with design load ration for each system during their own observation period.

The first observation is the COP of the referent systems is often higher the COP of the CO₂-based systems. This first figure is not a good one in order to compare each system because the load ration is different and the outdoor temperature too. First of all the next plot Figure 8-3 shows the COP of each system with a fix load ratio. This COP is called COP_LR3, because the fix load ration is equal to three.
Figure 8-3: Monthly average of the COP with a fix load ration equal to 3 for each system during their own observation period.

With the fix load ratio the shape of the curves are almost the same but the performance between each system change. For example RS3 has almost the same performance as RS1 while with the real load ration the performance of RS3 is the lowest of referent system. Nevertheless the CO₂-based systems get always a lower COP.

Normally the highest COP is during the winter when the outdoor is the lowest. However this trend is not followed by the system TR2. Indeed this system uses a heat recovery system in order to heat the supermarket during the winter. Therefore the condensing temperature is maintained around 25°C and the COP is lower than the others systems.

The condensing temperature
The condensing temperature depends of the outdoor temperature. In order to compare systems without this influence the COP_LR3 is plot in function of this condensing temperature.

The Figure 8-4 shows the monthly average of the condensing temperature for each system during their observation period. It seems obvious the condensing temperature decrease during the winter, however for TR2 the condensing temperature is higher during the winter than during the summer. The explanation is the TR2 system use the heat recovery to heat the supermarket during the winter. So voluntarily the choice of the condensing temperature is increase in order to permit the heat of the building. The choice is to the detriment of the performance of the system. On the Figure 8-3 the COP of TR2 is the lowest in winter contrary the others systems.
The majority of the systems have a floating condensing temperature, which follow the trend of the outdoor temperature. The only difference comes from the system TR2 which control the condensing temperature for the heat recovery during the winter. For the heat recovery this condensing temperature is maintained around 25°C.

Figure 8-5: COP with a fix load ratio for each systems compare to the condensing temperature.
The Figure 8-5 shows the monthly average of the COP with a fix load ratio (equal to 3) function of the condensing temperature. The graph permits to compare each system for the same condensing temperature. It appears two mains lines: the first one with the highest COP is the HFC-based systems and the second one with a lower COP is the CO2-based systems.

The linear regression permits to compare each system. The gradient means the influence of the condensing temperature over the system; for example more the gradient is low more the COP will decrease with the condensing temperature. The second coefficient is the intercept point; give information on the general performance of the system.

The HFC-based systems have almost the same coefficients of gradient and intercept point. TR1 has almost the same gradient compare to the HFC-based system, so the condensing temperature has the same on the performance. However the performance is generally lower.

TR2 has a gradient three times higher than the HFC-based systems. The condensing temperature has less effect on the performance, but this performance is the lowest of this five systems. Moreover the condensing temperature range is smaller than the others systems due the high condensing temperature during the winter for the heat recovery.

The Figure 8-6 shows the COP with a fix load ratio (equal to 3) function of the outdoor temperature. So the temperature range depends of the supermarket localisation.

![Figure 8-6: with a fix load ratio for each systems compare to the outdoor temperature.](image)

There are two lines like previously, however they are closer. The coldest systems are RS1 and TR1. The first observation is the CO2-based systems are less sensitive to the outdoor temperature because the TR1 gradient is two time higher than the others HFC-based systems. However the general performance is always lower.
TR2 has a gradient close to zero cause the heat recovery during the winter. However the heat recovery is turning off during the summer and the performance is close to the others HFC-based systems.

The two last figures (Figure 8-5 and Figure 8-6) don’t take into consideration of the evaporating temperature. And generally the evaporating temperature for the CO₂-based system has been chosen around -35°C for the freezer, which is lower than the HFC-based systems. This difference is to take in consideration in the comparison because it decreases the COP of the CO₂-based systems. Moreover the referent systems have the presence of the subcooling, which increase their performances.

The better way to compare the systems is to match this analyse with simulation in order to permit to show the influence of only one parameter.
9. **Bibliography**